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METHOD AND SYSTEM FOR COMPENSATING YAW RATE CHANGE BY LOAD CHANGE BEFORE OR DURING CORNERING

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Technical task:
Depending on their design (axle kinematics, tyre properties, engine behaviour, drive type), vehicles react differently to load change situations (change from train to thrust of drive force by releasing the accelerator pedal quickly) when cornering or directly before initiating cornering.
Furthermore, vehicles are increasingly being equipped with different vehicle dynamics control systems. Vehicle dynamics control systems that can actively generate a yaw moment (moment around the vertical axis of the vehicle) are increasingly being installed in production vehicles. Exemplary systems are:
- Superimposed differential (Quattro with sports differential) - Use of longitudinal slip effects on the tyre
- Brake torque vectoring (on the rear axle) - Use of longitudinal slip effects on the tyre
- Torque-Vectoring-Electric (on the rear axle) - Use of longitudinal slip effects on the tyre
- Active roll stabilisation - use of wheel load effects on the tyre

Further systems for influencing lateral dynamics can be found in active steering systems:
- Superimposed steering on the front axle
- auxiliary rear axle steering

Initial situation:
Due to a longitudinal force-related weakening of the front axle of front-wheel or all-wheel drive vehicles, a turning behaviour (reduction of the yaw rate) can occur when causing a load change situation in cornering or directly before initiating such a situation. This undesired lowering of the yaw rate thus leads to an increase in the curve radius and thus to understeering of the vehicle, which must be compensated for by the driver. Depending on the design of the vehicle, rear- or all-wheel drive vehicles may also experience turning behaviour (increase in yaw rate). Such an oversteer can bring the vehicle into an unstable driving state. Depending on the vehicle type and condition (load change, tyres), the load change reaction is individual and therefore difficult for the driver to predict and thus compensate.

Solution:
The yaw rate change of the vehicle caused by the yaw moment as a result of the load change situation can be compensated for with the aid of various chassis control systems by generating a counter-rotating yaw moment of corresponding height.
Requirement: The vehicle must have an active system(s) for generating yaw torque (e.g. superimposed differential, brake torque vectoring, torque vectoring electric, active roll stabilization) and/or an active steering system(s) on the front and/or rear axle.

In contrast to classic vehicle dynamics control systems such as the electronic stability program, a detected load change situation can be pre-controlled and thus compensated early with one or more active steering angle or yaw moment generating chassis systems. Such a control system can calculate an expected yaw rate deviation depending on the variables accelerator pedal position, applied drive torque, steering angle, vehicle speed, lateral acceleration and/or longitudinal acceleration, over which a compensating yaw torque is calculated (variant A, Fig. 0-1). For this purpose, a model of the relationship between the aforementioned variables and the expected yaw rate deviation or the required yaw rate is required.
compensating alloy torque, which can be based on empirical methods or vehicle models.
Furthermore, the compensating yaw moment can also be applied directly to a previously defined characteristic value for the characteristic of the load alternation situation by considering different combinations of the aforementioned variables (Variant B, Fig. 0-2). In this case, an attempt is made to simulate the yaw moment caused by load changes and ultimately invert it for compensation. For this purpose, various filter, delay and/or holding elements can be used to simulate the characteristic load change behaviour of the vehicle.

The yaw rate difference or yaw torque can be compensated using one or more yaw torque and/or steering angle actuators. The corresponding formulae are also derived from the linear single-track model in the form of the transfer functions of front axle steering angle δv, rear axle steering angle δh and/or yaw moment Mz to yaw rate Ψ˙ from the oblique stiffness at front c_Sv and rear axle c_Sh, the vehicle mass m, the longitudinal speed v and the wheelbase l. The conversion of a yaw moment into a steering angle takes place on the same basis:

$$\left(\frac{\dot{\Psi}}{\delta_v}\right) = \frac{v}{l + \frac{m}{l} \cdot \left(\frac{l_h}{c_{Sv}} - \frac{l_v}{c_{Sh}}\right) \cdot v^2}$$

$$\left(\frac{\dot{\Psi}}{\delta_h}\right) = \frac{v}{l + \frac{m}{l} \cdot \left(\frac{l_h}{c_{Sv}} - \frac{l_v}{c_{Sh}}\right) \cdot v^2}$$

$$\left(\frac{\dot{\Psi}}{M_z}\right) = \frac{c_{Sv} + c_{Sh}}{c_{Sv} \cdot c_{Sh} \cdot l} \cdot \frac{v}{l + \frac{m}{l} \cdot \left(\frac{l_h}{c_{Sv}} - \frac{l_v}{c_{Sh}}\right) \cdot v^2}$$

The distribution of the yaw rate deviation over several systems (Figure 0-3) can take place in different ways. On the one hand a parallel distribution (1) to several systems is possible, which carry the intervention in defined portions (2). On the other hand, it is possible to perform the distribution hierarchically. In this case the systems are evaluated regarding their effectiveness for the compensation of the yaw rate deviation in the appropriate situation and the intervention is then distributed sequentially to the systems, whereby the requirement is distributed to the systems in order of descending effectiveness. Once a system has exhausted its potential, a transfer (3) to the next more suitable system takes place. A combination of both arbitration strategies is also possible.

Using one or more active systems, the individual control variables δv, δh as well as Mz. are then obtained for the systems front axle steering nδv, rear axle steering nδh and yaw torque adjuster nMz, taking into account the previously determined components:

$$\delta_v = n_{\delta v} \cdot \frac{\Delta\dot{\Psi}}{\left(\frac{\dot{\Psi}}{\delta_v}\right)}$$

$$\delta_h = n_{\delta h} \cdot \frac{\Delta\dot{\Psi}}{\left(\frac{\dot{\Psi}}{\delta_h}\right)}$$

$$M_z = n_{Mz} \cdot \frac{\Delta\dot{\Psi}}{\left(\frac{\dot{\Psi}}{M_z}\right)}$$

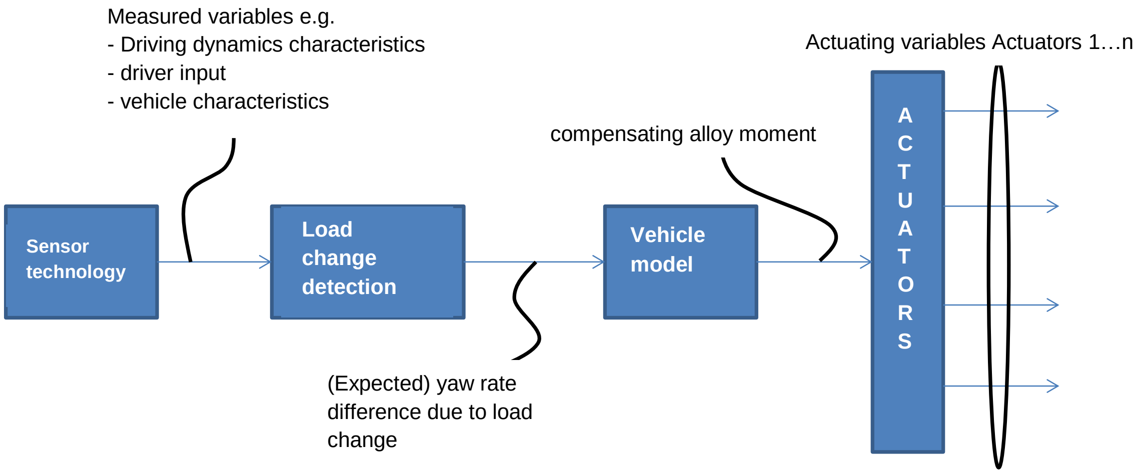

Figure 0 1: Architectural design and signal characteristics for compensation of the yaw rate deviation from the desired behavior caused by the load change reaction by model-based consideration (variant A).

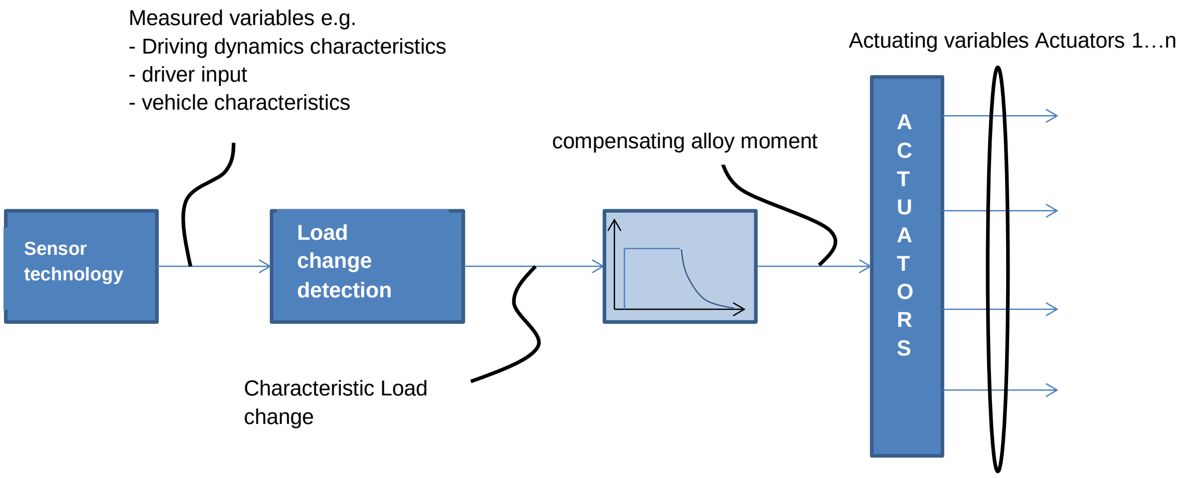

Figure 0 2: Architectural design and signal characteristics to compensate for the yaw rate deviation from the desired behavior caused by the load change reaction by simulating and inverting the resulting yaw torque.

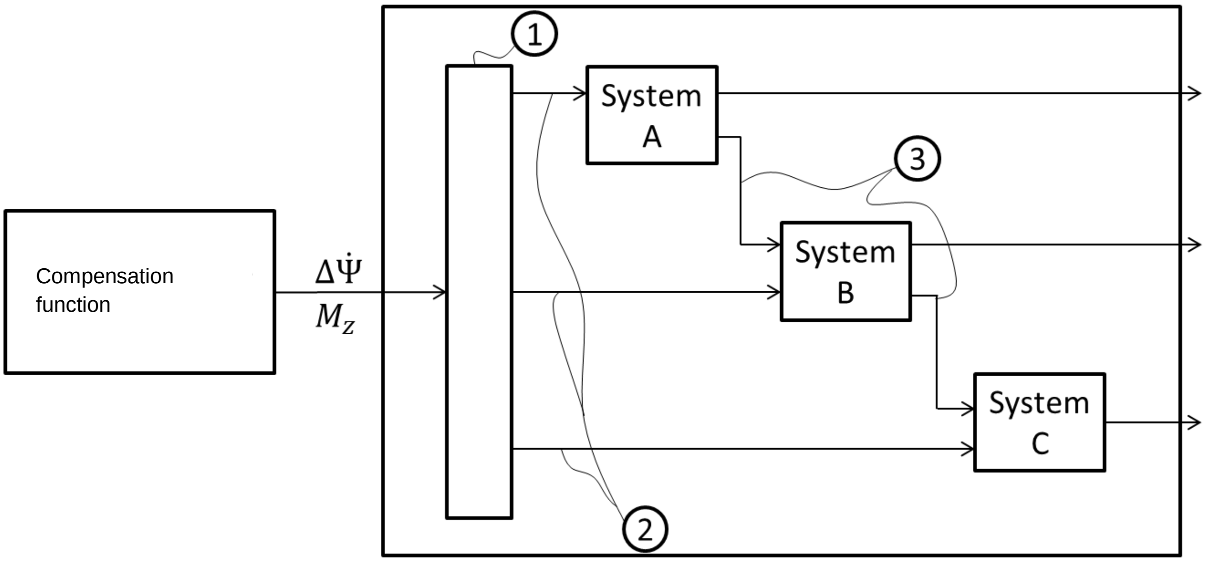

Figure 0 3: Function structure

Advantages:
Compensation of the yaw rate change of the vehicle caused by the load change reaction, thus maintaining neutrality and predictability of the vehicle reaction.